Energy optimization in ice hockey halls I. The system COP as a multivariable function, brine and design choices.

A. Ferrantelli[a], P. Melóis[b], M. Räikkönen[a], M. Viljanen[a]

[a] Department of Civil and Structural Engineering, P.O.Box 12100, FI-00076 Aalto, Finland
[b] Institut National des Sciences Appliquées de Toulouse, 31077 Toulouse Cedex 4, France

Abstract

This work is the first of a series of articles addressing the energy optimization in ice hockey halls. Here we outline an analytic method to predict in which design and operating conditions the COP of the entire cooling system (refrigerator and cooling tower) COP_{sys} is maximum.

COP_{sys} is investigated as a function of several variables, like electric consumption and brine physical properties. With this method, the best configuration and brine choices for the system can therefore be determined in advance.

We estimate the optimal design of an average-sized ice rink, including pipe diameter, depth and brine type (ethylene glycol and ammonia). We also single out an optimal brine density and show the impact of the electric consumption of the pump on COP_{sys}.

Our theoretical predictions are validated with heat flow measurement data obtained at an ice hockey hall in Finland. They are also confronted with technical and cost-related constraints, and implemented by simulations with the program COMSOL Multiphysics.

The multivariable approach here introduced is general, and can be applied to the rigorous preliminary study of diverse situations in building physics and in other areas of interest.

Keywords: energy saving, sustainability, system optimization, theoretical models

Contents

1. Introduction

Indoor ice rinks are large buildings where an ice sheet (of area between $1000\,m^2$ and $2300\,m^2$) is cooled and kept at a temperature lower than -5^oC by a complex refrigeration system, that includes compressors, brine pumps, evaporators and condensers.

Clearly, the above implies a very high energy consumption, to which one should add also the heating of spectator stands and the air conditioning. A major challenge is then given by the implementation of methods for saving energy and reducing operational costs.

This task can be accomplished in several ways. For instance, literature on mathematical models for heat transfer in the concrete slab has recently flourished [1, 2, 3, 4, 5, 6], thanks also to the fast development of computational tools.

However, the above mentioned research focuses on a precise study of heat transfer in the concrete slab, and/or on the ice surface. The results give a very precise analysis of the phenomenology of heat transfer, but do not manifestly provide with indications on how the ice hockey hall should be designed in order to save energy.

Here we adopt a fairly different (yet complementary) approach, that aims to act toward this direction. The basic idea is the following: focus on the quantities of interest

and investigate their behavior in function of design parameters[1]. This gives an optimal theoretical configuration, through a quick and simple mathematical analysis. After considering technological constraints and costs, one eventually gives practical prescriptions on how the system should be improved (e.g. best brine, pipe size etc.).

This prescription allows to single out the best design configuration of a system *a priori*, still at the design stage. By means of a quick mathematical study, which reduces to a simple maxima/minima study of a multivariable function, one is able to predict the best design configuration without e.g. expensive empirical tests.

This basic idea is so general that it can be used not only in any heat transfer processes, but also in any areas of interest to engineers. Moreover, as we show in this paper, it is certainly complementary to the accurate mathematical studies presented in e.g. [7, 8, 9, 10]. A detailed knowledge of the physical processes can and should be used indeed for enhancing the predictive power of the model.

In this article we provide a specific example of this generic procedure, that is applied in this case to the Coefficient Of Performance (COP) of the *entire* cooling system (refrigerator and cooling tower). This COP_{sys}, here viewed as a multivariable function of the system parameters, such as pipe length and brine density, can be maximized by an appropriate choice of such parameters.

A low efficiency of the cooling system (low COP) is a primary source of energy and money waste. Therefore the challenging topic of energy saving should include maximization of the COP. It is clear from the above that this reduces to simply studying the impact of each design parameter on the COP magnitude, thus finding the best configuration.

We do not consider only the COP of the compressor because of the bound on the evaporation temperature ($\sim -15\,^oC$) existing in ice hockey halls. The condensation temperature can be indeed changed[2] according to the season and weather conditions. This cannot occur for the evaporation temperature, therefore there is not much freedom in the optimization of the compressor COP. We can instead circumvent this limitation by considering the *full* cooling system. This also constitutes a non-standard approach in this kind of analysis.

Since the problem of energy saving in such a complex system as an ice hockey hall has several options, our study as a whole includes a number of perspectives divided into several articles. Here we address only the case of the COP as a multivariable function of the design parameters.

The present paper is structured as follows.

Section 2 discusses briefly the experimental setup and the measurements.

In Section 3, we consider the coefficient of performance COP_{sys} of the cooling system as a multi-variable function of the geometry of the ice rink, electric consumption and physical properties of the brine. We show an excellent consistency between predicted and measured COP_{sys}.

In Section 4, the model proves that COP_{sys} is much more dependent on the electric consumption of the cooling tower (related to condensation and evaporation temperatures) than on the pumping power. This agrees with known experimental results.

In Section 5, we use our theoretical model to compare COP_{sys} values according to

[1]In this paper, the function is COP_{sys} and the variables are pipe size, brine density etc.

[2]This is done in the ice hockey hall in Leppävaara, where we use the condensation energy available.

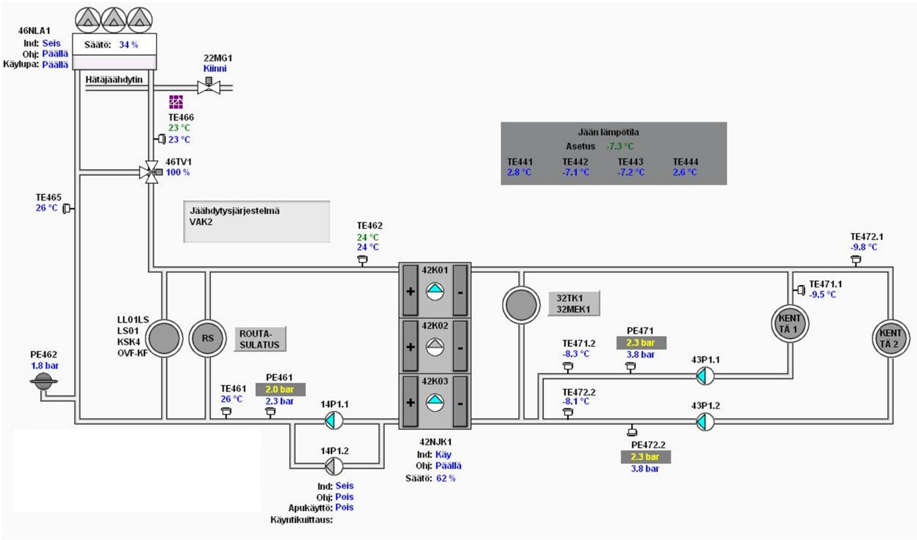

Figure 1: The Leppävaara ice hockey hall (three compressors in the middle, "kenttä 1" and "kenttä 2" are the two ice tracks) [11].

different ethylene glycol concentrations and ammonia. We show that a more diluted ethylene glycol solution, or ammonia, are preferable.

In Section 6 we focus on the design optimization, via simulations of heat transfer in the concrete slab with the program COMSOL. We show that the depth and size of the pipes inside the slab are not crucial to the heat transfer process. However, increasing the pipe number by 1/3 is an option which could be of interest in the analysis of control of the brine flow.

In Section 7 the technological constraints of our theoretical predictions, based on the validated theoretical model, are discussed together with an estimation of costs.

2. Ice hall structure and measurements

In this paper, we consider a theoretical model based on measurements taken at an ice hockey hall built in 2009 in Leppävaara, Espoo, Finland (see Fig.1). It contains two ice tracks.

Heat flux and temperature are measured with a plate installed at the ice-concrete slab interface. The sensors were placed about three meters from the rink so that during the resurfacing only one layer of warm water is spread over the heat flux plate.

The heat flux plate and thermal sensors are connected to a logger, which records an input sent by the sensors every five minutes. Thus we get instantaneous values of the interface temperature in [C], and of the heat flux from the ice pad in [W/m^2], by multiplying the values saved in the logger by the individual calibration factor of the plate (see Fig.2). As we know the surface area of the ice track, we can calculate the total heat rate in [W].

In order to keep the ice frozen, the heat gains have to be canceled by the cooling power. This is generated with a refrigerating machine located in a separate container,

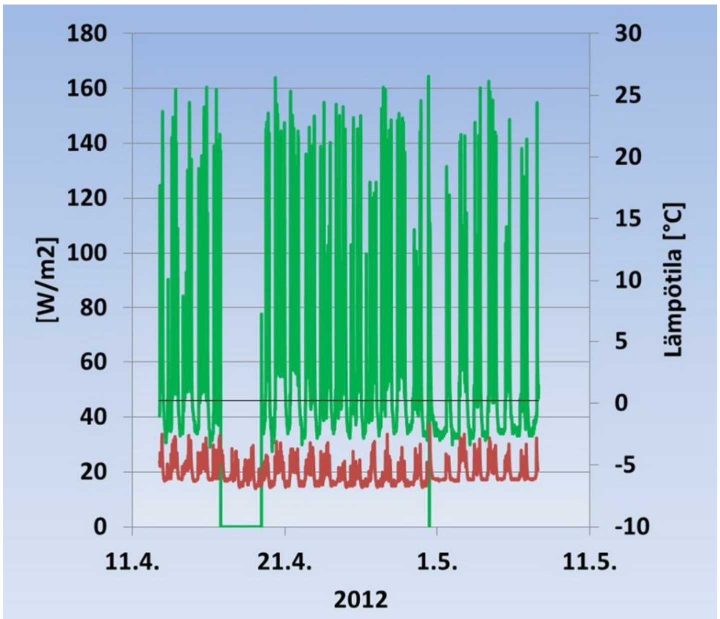

Figure 2: Heat flux and interface temperature in the ice track n.1 in the Leppävaara ice hockey hall [11].

in a backyard of the main building. The electric consumption of the whole refrigerating machine including compressors, brine pumps, evaporators and condensers is measured by a pulse meter, which is connected to a logger set to save the cumulative value of the pulses received every five minutes (Fig.3). Again by multiplying this value we find the amount of used energy in [J], or dividing by this interval, the average power [W] during the period of measurement.

The design power of the refrigerating machine is 610 kW. Refrigeration is guided by the thermal sensors of the ice track. The evaporation temperature is -15^oC and the condensation temperature is between 24^oC and 26^oC. There are three compressors, and the circulating refrigerant is NH_3, otherwise ammonia. The brine pump for the ice track has electricity power 11 kW and the flow is $28\,l/s$. The brine used is an ethylene glycol solution at 35%. The power of the brine pump for the condensate is 7.5 kW. There are 10 condensers[3], and the power of the condenser fan is 3.2 kW.

3. The COP as a multi-variable function

Optimization of a cooling system can be achieved in several ways. Here we concentrate on maximizing the Coefficient Of Performance (COP) of the *entire* cooling system. It is defined as

$$\mathrm{COP}_{sys} = \frac{\text{cooling power}}{\text{electric consumption}} = \frac{Q}{W_p + E}\,, \tag{1}$$

[3]This class of cooling systems is very well described in [12].

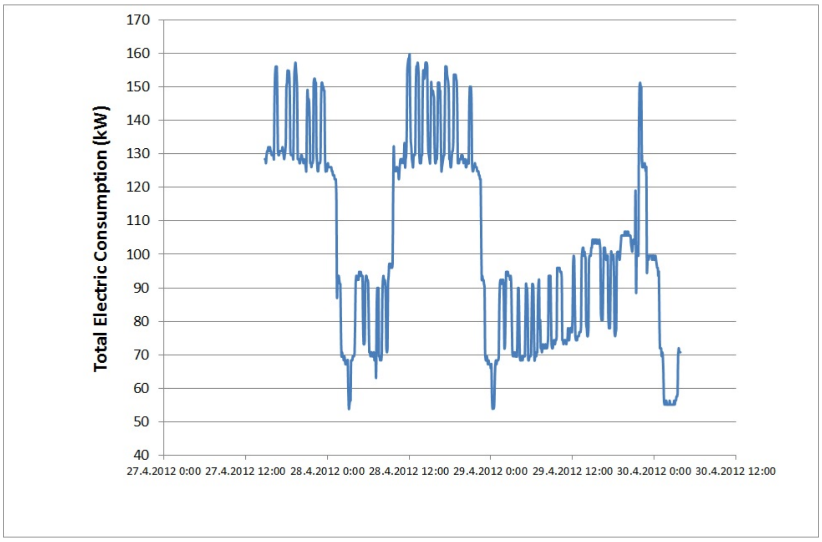

Figure 3: Total electric consumption $E_{tot} = W_p + E$, as measured in four days.

where we have split the energy consumption in the two contributions of pumping power W_p of the brine pump and electric consumption E. The latter comprises the power consumed by the refrigeration system described in Section 2, namely condensers and compressors.

W_p and the cooling power Q are instead functions of the volumetric flow $\dot{V}$, brine temperature difference, geometry of the cooling system (length and diameter) and brine physical properties.

Let us now expand the expressions for the cooling power Q and the pumping power W_p, to rewrite the COP in function of the variables of interest. First, expand the cooling power:

$$\mathrm{COP}_{sys} = \frac{Q}{W_p + E} = \frac{\dot{m} c_p \Delta T}{W_p + E} = c_p \rho \frac{\dot{V} \Delta T}{W_p + E} \,. \tag{2}$$

Then, the total pressure drop is the sum of contributions from the collector pipes and from the pipes under the ice rink (subscript p). The pumping power can thus be written in the following form,

$$W_p = \dot{V} \Delta p = \dot{V} \left(f \frac{L}{D} \frac{\rho}{2} v_m^2 + f_p \frac{l}{d} \frac{\rho}{2} v_{m,p}^2 \right) = 0.14 \rho \dot{V}^2 \left(\frac{L \nu^{1/5}}{D^{24/5} \dot{V}^{4/5}} + 1.91 \frac{l}{d^4} \nu \right) \,, \tag{3}$$

where L and D are resp. the length and diameter of the main collector pipes, l and d are those of the 150 pipes under the ice rink and v_m, $v_{m,p}$ are the mean bulk velocities of the brine in the collector and ice rink pipes, respectively[4]. f and f_p are the corresponding friction coefficients. It can be shown that, for our average volumetric flow $\dot{V} = 37.57\, l/s$,

[4] Remember that $v_m = \dot{V}/A$ and $v_{m,p} = \dot{V}_p / A_p = \dot{V}/(150 A_p)$

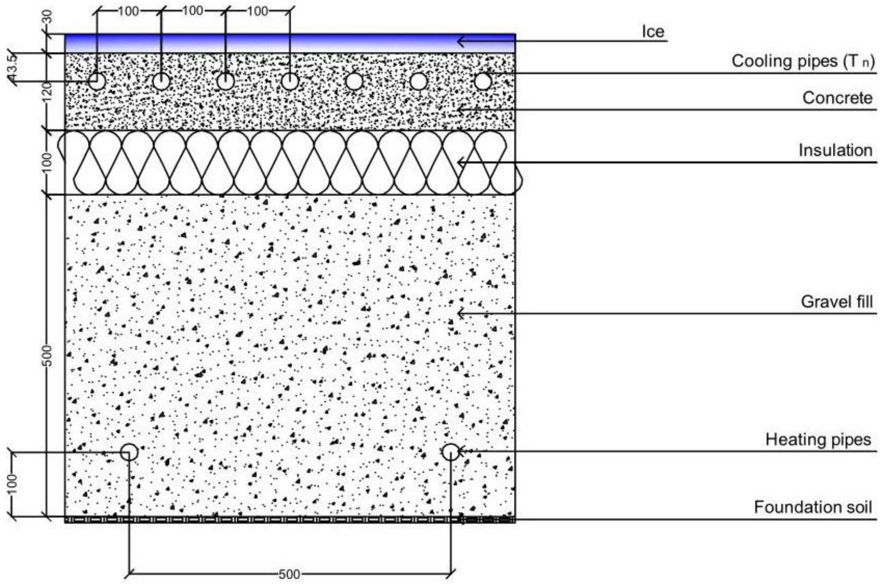

Figure 4: Ice pad construction at the Leppävaara ice rink.

the fluid motion in the collector pipes is turbulent (Re $\sim$ 35000), while in the smaller pipes it is laminar (Re = 1052). This justifies the exponentials in Eq.(3).

By substituting the above expression for the pumping power in (2) and simplifying, we get our formula:

$$\mathrm{COP_{sys}} = \frac{c_p \dot{V} \Delta T}{\left(0.14 \frac{L}{D^{4.8}} \nu^{0.2} \dot{V}^{2.8} + 0.27 \frac{l}{d^4} \nu \dot{V}^2\right) + \frac{E}{\rho}} \,. \tag{4}$$

Therefore one can express the COP implicitly as follows:

$$COP = f(\dot{V}, \rho, \nu, c_p, L, d, E) \,, \tag{5}$$

namely by considering the COP as a function of the above variables, one could investigate the impact of each of them on the energy optimization.

Finding for which values of these parameters the function COP is maximal corresponds to choosing the best possible configuration of brine fluid, volumetric flow, electric consumption and so on. We are therefore applying to our problem the method of *functional optimization*, which is used in a number of diverse research topics [13, 14, 15, 16, 17, 18, 19, 20].

In the next sections, we will take into consideration each and all of the variables entering (4), and calculate for which values the function (5) is maximal.

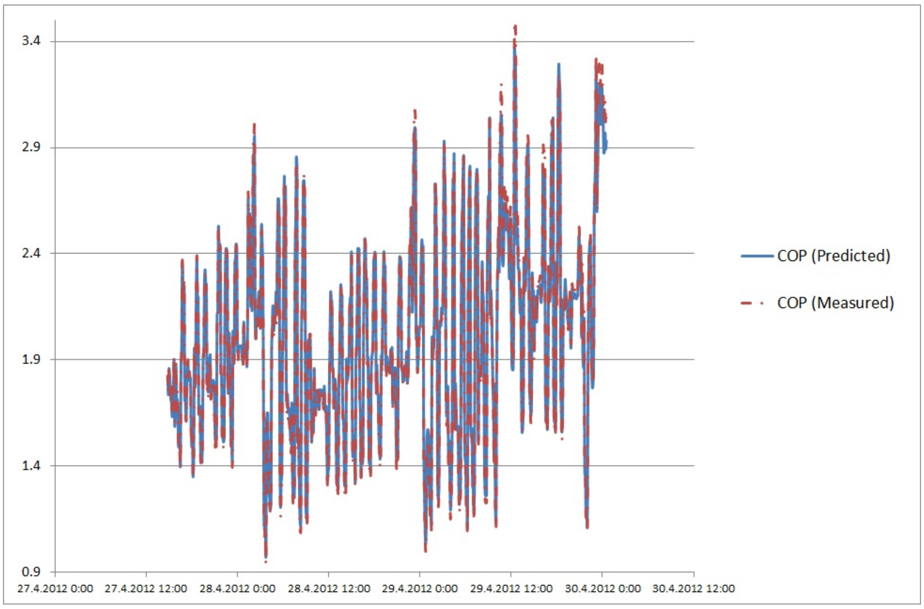

Figure 5: Predicted COP, computed with Eq.(4), vs the COP as measured at the ice rink.

4. Relative weight of pumping power and electric consumption in COP$_{sys}$

We first focus on the relative impact of W_p and E on the COP. We will see that this analysis is crucial for energy optimization, and that it leads also to the validation of Eq.(4) with the experimental data.

In Eq.(2), the denominator $W_p + E$ corresponds to the *total* electric consumption of the compressor,

$$E_{tot} = W_p + E\,, \tag{6}$$

that is measured at the ice rink in Leppävaara. This is, on the average, of order $E_{tot} \sim 120kW$ (recall that the measurements were taken every 5 minutes).

Since the data do not contain values of the pumping power and E separately, we compute $\dot{V}$ from the measured values of ΔT and from the cooling power Q. Substituting the according pumping power into (4) and comparing the COP thus obtained with the measured value, we then find E as a fraction of the measured E_{tot}.

The above method returns the plot shown in Fig.5, showing an excellent agreement between data and theory and implying that E corresponds to 93.4% and W_p to 6.6% of the total electric consumption E_{tot}. This is consistent with well-known results (see for instance [21]).

4.1. Impact of the volumetric flow on COP$_{sys}$

Consider Eq.(4) as a function of only the volumetric flow $\dot{V}$ and the temperature difference of the brine at inlet and outlet, ΔT.

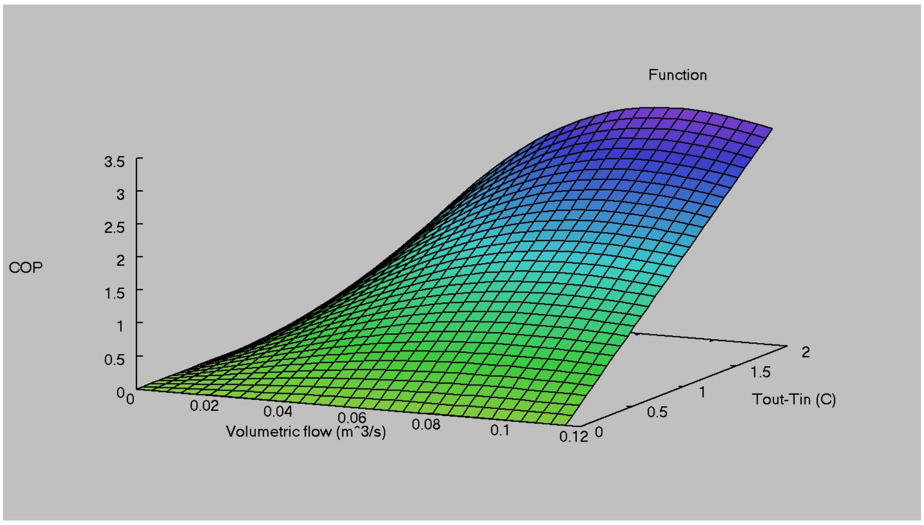

Figure 6: COP_{sys} as a function of volumetric flow and brine temperature difference at inlet and outlet.

COP_{sys} is clearly proportional to ΔT, while the dependence on the volumetric flow is nontrivial. Thus we wonder whether there is a specific value of $\dot{V}$ such that the ratio cooling power vs pumping power reaches a maximum, leading to maximum COP[5].

The COP can be plotted in a 2-dimensional graph as in Figure 6, and as a function of only $\dot{V}$ if ΔT is fixed. We show this for $\Delta T = 1.5^o C$ in Figure 7, where one can see that the COP increases until it reaches a maximum for a specific value of the volumetric flow, before decreasing again.

The maximum corresponds to $\dot{V} \sim 90\, l/s$, that is a very large value. However, this is expected as we have just demonstrated that the pumping power weights for only $\sim 7\%$ of the total electric consumption. In other words, we have shown that in principle, it is possible to reach the maximal COP_{sys} by increasing the pumping power to very high values.

Therefore we conclude that changing the magnitude of W_p, or in other words, working on the brine pump, is not determinant for energy optimization. This happens since technological limits to $\dot{V}$ do exist, and we have shown that the pumping power constitutes a small portion of the total electric consumption.

Rather, looking at Eq.(4) for the COP, our interest should be directed towards the following:

1. Reduction of the electric consumption E, which comprises the chiller, condensers and heat exchanger.
2. *Control* of the brine flow, in response to environmental factors on the ice rink, such as resurfacing and skaters. The aim is to minimize the (lowest) peaks in Fig.5 and obtain a smoother shape of COP_{sys}.
3. Choosing the most efficient and cost-balanced brine for our configuration.

[5]Remember that E is independent of $\dot{V}$.

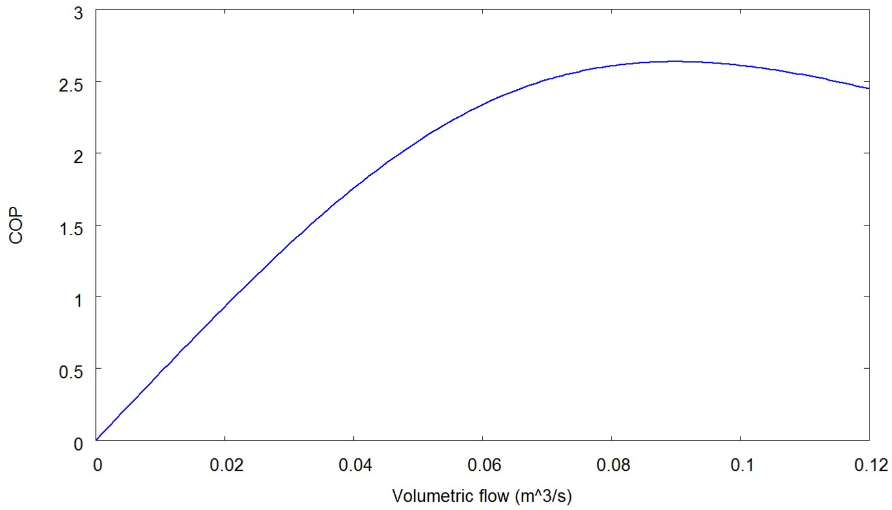

Figure 7: COP_{sys} as a function of only the volumetric flow.

4. Verifying if it is possible to get an optimal design configuration (e.g. pipe diameter and number), to maximize the cooling efficiency.

Points 1) and 2) require a dedicated analysis, therefore they will not be investigated here. We will address them in the forthcoming papers. In the next sections we focus instead on points 3) and 4).

5. Brine choices

First of all, recall Eq.(4), namely

$$\mathrm{COP}_{sys} = \frac{c_p \dot{V} \Delta T}{\left(0.14 \frac{L}{D^{4.8}} \nu^{0.2} \dot{V}^{2.8} + 0.27 \frac{l}{d^4} \nu \dot{V}^2\right) + \frac{E}{\rho}} \, .$$

We want to see how the specific brine chosen is relevant for the COP. Let us study then how the above expression is dependent on the physical characteristics of the fluid.

We have already shown that the pumping power counts for only $\sim 7\%$ of the total electric consumption, thus the kinematic viscosity ν is irrelevant. One should then focus on the density ρ and the specific heat c_p. Since

$$\mathrm{COP}_{sys} \propto c_p \, , \, \mathrm{COP}_{sys} \propto \frac{\rho}{E} \, , \tag{7}$$

and in general $c_p \propto 1/\rho$, it is not trivial to decide which parameter should be increased to achieve higher COP. One then needs to write ν and c_p in function of ρ, to obtain a plot COP_{sys} vs ρ which singles out the point of maximum.

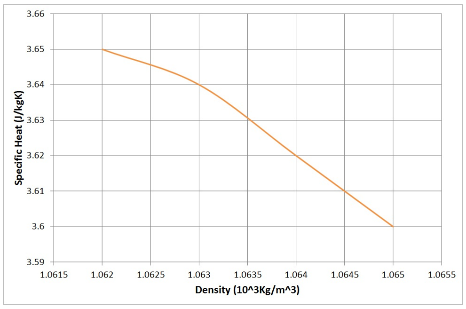

Figure 8: Specific heat of an ethylene glycol solution at 34% in function of density.

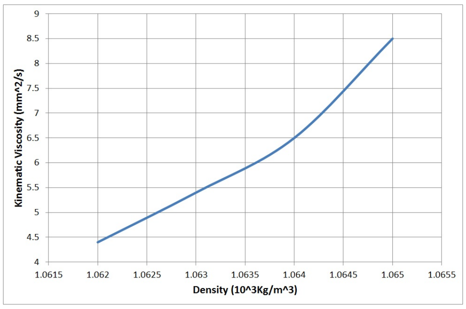

Figure 9: Kinematic viscosity of an ethylene glycol solution at 34% in function of density.

However, there are no theoretical formulas $\nu, c_p = f(\rho)$, therefore we interpolate experimental data taken at our laboratory, for an aqueous solution of ethylene glycol at 34%. We obtain the following relations,

$$\nu(\rho) = 24725\rho^2 - 524575\rho + 278244\,, \tag{8}$$
$$c_p(\rho) = -2517.5\rho^2 + 5337.6\rho - 2825.54\,. \tag{9}$$

These provide with estimates which are sufficiently accurate for our purpose. The corresponding experimental curves are reported in Figs.8 and 9. For typical values of our ice rink configuration ($\dot{V} = 37\,l/s$, $\Delta T \sim 1.5^oC$, $E \sim 90\,kW$), we then get the COP as a function of only the brine density as in Fig.18.

Last, let us study how COP$_{sys}$ changes according to the average brine temperature $(T_{in} + T_{out})/2$. We use again our experimental data to derive the following interpolation formulas,

$$\rho(T) = 1.1176 - 0.0002T\,, \tag{10}$$
$$\nu(T) = 0.01T^2 - 5.678T + 810.08\,, \tag{11}$$
$$c_p(T) = -0.0001T^2 + 0.0575T - 4.6061\,. \tag{12}$$

To increase the precision of the fit[6], we consider the temperature interval of our interest, $263\,K < T < 278\,K$. The above equations are plotted in Fig.25, Fig.26 and Fig.27. By plugging these expressions into Eq.(4), we obtain the COP values in function of the average brine temperature, for typical values of our ice rink configuration ($\dot{V} = 37\,l/s$, $\Delta t \sim 1.5^oC$, $E_{tot} \sim 90\,kW$), as in Fig.21. This agrees with the experimental COP values.

5.1. Ethylene glycol concentration

Let us go further with the study of the COP dependence on the brine characteristics according to Eq.(4). Here we aim to understand whether we can generalize the results of the previous section. To do this, we compare the values of COP$_{sys}$ obtained for four different ethylene glycol concentrations, with the same system configuration (volumetric flow, electric consumption etc.).

[6]These fits are precise enough for our qualitative study

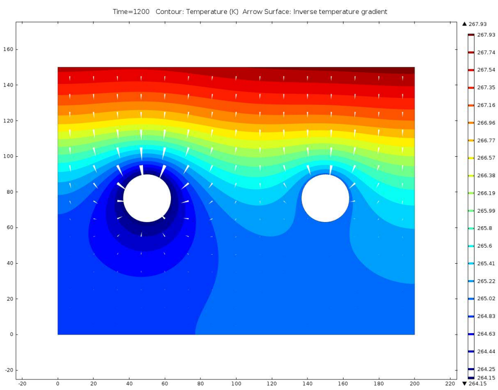

Figure 10: Simulation of heat transfer with two pipes.

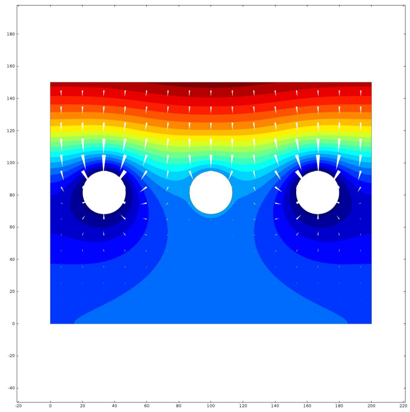

Figure 11: Simulation of heat transfer with three pipes.

The results are plotted in Fig.21: it is clear that the lower the concentration, the higher the COP. To generalize this, we consider the plots in Fig.22, Fig.23 and Fig.24, which concern respectively the specific heat, density and kinematic viscosity of ethylene glycol in function of the concentration, at $T = -10^o C$.

By comparing these graphs with the COP in function of the concentration, we see that both ν and ρ increase, while c_p decreases. However, COP$_{sys}$ behaves like c_p. Accordingly, we conclude that, at least for an ethylene glycol brine, *the specific heat is the most crucial physical parameter for optimizing the system COP.*

5.2. Ethylene glycol or ammonia?

One last issue to take care of, is which brine would be better for maximizing COP$_{sys}$. We consider an aqueous solution of 14% ammonia (NH_3) and calculate the according COP with (4); the result is then compared to the COP with ethylene glycol ($C_2H_6O_2$) at 20% and 34%. We find the following,

$$\mathrm{COP}_{sys}(NH_3) > \mathrm{COP}_{sys}(C_2H_6O_2)\,, \tag{13}$$

for any ethylene glycol concentration. In particular, for a solution at 20% the gain is 9%, whilst for a solution at 34% the COP is larger by ∼18%, that is quite remarkable. We conclude that according to our analysis, ammonia should be preferred to ethylene glycol.

6. Design optimization

We now address the question of how are the pipe number, pipe diameter and pipe depth in the slab relevant to the heat transfer. In other words: is it possible to obtain a more efficient ice cooling by altering any of these design parameters?

In this case simulations are very handy, as they do not require expensive and time-consuming experiments, and provide already with good indications for further investigations, if those are found to be necessary.

Let us consider the two-dimensional slab of Fig.4. Following its symmetry, we focus on a 200 mm wide section, containing two pipes with 100 mm separation, and perform simulations of conduction heat transfer with the program COMSOL Multiphysics (Fig.10).

The setup is the following: the concrete slab has dimensions 200x120 mm, the pipe external diameter is 27 mm, the reference depth (at the pipe's edge) is 30 mm and the ice thickness is 30 mm as well. We used densities $\rho = 2000\,kg/m^3$ for concrete, $\rho = 918.9\,kg/m^3$ for ice and $\rho = 1.066\,kg/m^3$ for a 34% ethylene glycol solution.

A uniform heat flow $q = 60\,W/m^2$, consistent with our measurements, is applied to the ice surface, at initial temperature $T_i = -4^oC$. Time-dependent heat transfer occurs between the ice and the pipes, at temperatures $T_{in} = -9^oC$ and $T_{out} = -8^oC$ for the inlet and outlet pipe. These are the average tube temperatures in our data. The following results are obtained after $t = 1200\,s$.

6.1. Changing the pipe diameter

First we consider different pipe sizes: $d = 27\,mm$, $d = 35\,mm$, $d = 40\,mm$. The latter diameter is certainly too large, but it is useful for visualizing more clearly the trend of the temperature vs the pipe size.

Our results for the temperature at the ice surface and ice/concrete interface are plotted in Fig.19. We find that, for an increase in the diameter from 27 to 40 mm, namely 148%, the temperature changes by only $\sim 1\%$ (see Table 1). Moreover, the temperature profiles do not change qualitatively, as shown by comparing Fig.14 for $d = 27\,mm$ and Fig.12 for $d = 40\,mm$. We conclude that the pipe size has a very small role in the optimization of the heat transfer. This is clearly due to the high heat capacity of ice and concrete.

6.2. Changing the pipe depth

The same method has been used when changing the pipe depth in the range $25\,mm < h < 30\,mm$. Our results are reported in Table 1 and in Fig.20. Also in this case, the system is not very responsive to a large increase of the parameter (here, the depth).

From Table 1 we see indeed that pipes closer to ice by 20% lower the temperature by only $0.1\,K$. Comparison of Fig.13 and Fig.14 shows also that the temperature profiles are qualitatively the same. Therefore the pipe depth is basically uninfluent to cooling optimization.

6.3. Two versus three pipes

Depth (mm)	$T_{ice}(K)$	$T_{con}(K)$
25	267.97	266.74
27	267.99	266.78
30	268.03	266.86

Diameter (mm)	$T_{ice}(K)$	$T_{con}(K)$
27	268.03	266.86
35	267.97	266.70
40	267.91	266.59

Table 1: Maximum temperature of the ice surface and ice/concrete interface for different pipe depths and diameters.

Consider the 200 mm wide slab in Fig.10, with one inlet and one outlet pipe. We wonder how the heat transfer is modified by adding one more inlet pipe in the same

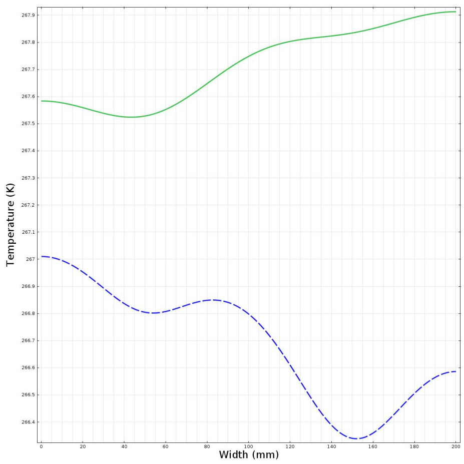

Figure 12: Temperature profile at the ice surface (solid) and ice-concrete interface (dashed), for two pipes with diameter $d = 40\,mm$.

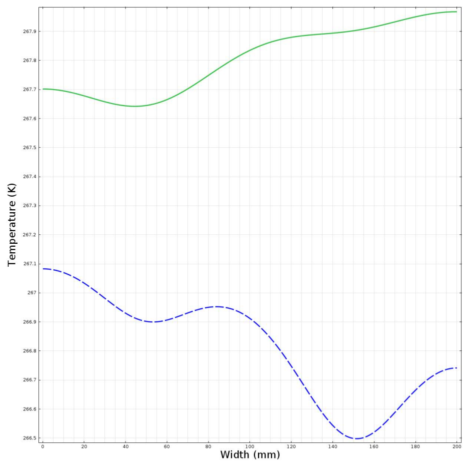

Figure 13: Temperature profile at the ice surface (solid) and ice-concrete interface (dashed), for two pipes buried at depth $h = 25\,mm$.

width, thus installing the pipes closer to each other. This is visualized in Fig.11: the distance between two contiguous pipes is now changed from 100 mm to 66 mm.

COMSOL simulations show that the temperature profiles at the ice surface and ice/concrete interface are radically different in the two cases (compare also Figs.14 and 15). Besides the general lowering of T values at a specific x, the temperature profiles are much smoother and the heat transfer is more uniform for three pipes.

Using 200 pipes instead of 150 for the whole ice rink could then be an interesting option. We see from Figs.14 and 15 a remarkable difference of $T(x)$ at the ice surface. The temperature at $x = 200\,mm$ is almost 0.5^oC smaller for three pipes, and its profile is much more uniform than in the case of two pipes. Moreover, with three pipes the whole system is clearly more responsive to temperature changes. This might be relevant when addressing control techniques of the brine flow, and will be taken into consideration in a further study.

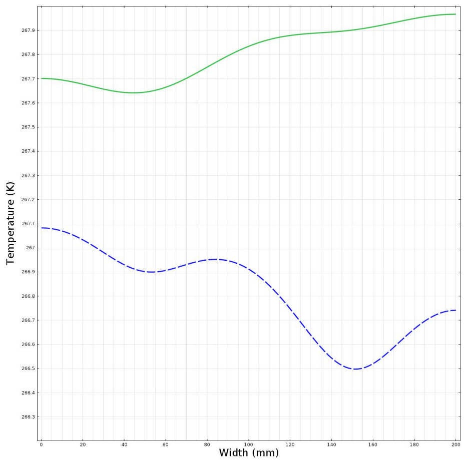

Figure 14: Temperature profile at the ice surface (solid) and ice-concrete interface (dashed), in the case of two pipes.

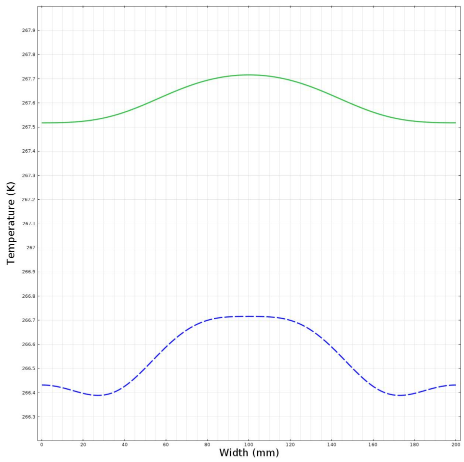

Figure 15: Temperature profile at the ice surface (solid) and ice-concrete interface (dashed), in the case of three pipes.

7. Technological constraints and cost estimate

The previous sections have provided with a series of insights and prescriptions for optimization, which are based on the functional method we outlined (top-down approach). These are validated with measurements performed at an ice hockey arena.

This section aims to complement such predictions with a bottom-up approach, discussing their practical limits from the viewpoint of technical design and costs. We refer here to several sources, see for instance [22, 23, 24, 25].

7.1. Pipe depth

The refrigerating slabs contain pipes buried in concrete, therefore it is crucial to know the minimum depth of installation for the pipes under the upper surface of the concrete slab.

There are several limitations on the thickness of the concrete layer. It must resist to a certain pressure (users, ice resurfacers), it works in compression and flexion. It should resist to fatigue, since several cycles of loading and unloading occur each day. The concrete must resist to multiple cycles of freezing and unfreezing, and must not let humidity reach the insulation layer (proper porosity), to avoid moisture-related problems.

Some regulations set the minimum thickness of these types of slabs to 130mm, but these are for industrial applications (warehouses, sheds etc.). In our case, the maximum load is inferior to $10\,kN/m^2$ (distributed or concentrated) so the thickness of the slabs can be undersized.

For an 80mm to 100mm polystyrene insulation layer (that can resist to maximum $850\,kN/m^2$), the minimum thickness of the concrete slab should be 65mm to resist to the maximum vertical load. Usually, this size goes up to 120mm in order to accommodate the pipes. Some North-American ice-rinks designs give us a thickness of 6in (150mm)

which is oversized. Moreover, at least one layer of wire mesh reinforcement is usually placed at the concrete-ice interface.

Overall, the above limitations lead to design the concrete layer with a minimum concrete thickness of 25mm *above* the edges of the pipes.

7.2. Slab design and pipe size

Different formulations of concrete can be used, like hemp concrete. A wire mesh reinforcement is needed especially if the arena is used for other purposes after the skating season is over, because the floor has to withstand a large spectrum of average load. In this case a strict regulation is to be applied to determine the distance between the steel wires and thus the layer thickness. One might also need to consider the heat transfer through this material.

Note that when designing the concrete slabs, the stress and dilatations to which the pipes are submitted because of the variation of temperature should be taken into account as well.

The PVC pipes are built to resist the pressure of the concrete layer above. In our case, a 27mm to 40mm diameter pipe will resist a pressure of a concrete layer with thickness from 25mm to 130mm (weight of the concrete plus steel reinforcements).

Concerning the maximum depth of the pipes, the thickness of concrete *under* the pipes must not be too small to avoid direct contact with the insulation layer. However, as long as this contact is avoided there are no limitations. Placing the pipe in the middle of the concrete layer is a good solution (easy to set up). Note that another solution consists of prefabricating the slabs and seal them together on site, but this makes it more difficult to install a perfectly flat surface.

7.3. Price of the PVC Pipes

The study performed in this paper is based on a 200mm wide slab containing 2 or 3 pipes spaced out 100mm or 66mm each. Let us consider an Olympic ice rink of 30m width, for a total of 300 or 450 pipes. Then the total length of the PVC pipes would be 17 000m or 27 000m (60m per pipe). Of course installing more pipes (3 per slabs rather than 2) will be more expensive, even if we chose the minimum diameter of pipes (the price-diameter relation is non-linear). What must also be taken into account is the price of the 180° bend connecting two parallel pipes. For N pipes we need N-1 bends.

The average price for a 25mm to 40mm diameter PVC pipe for refrigerant applications is approximately 0.20€ to 0.50€per meter (industrial supplier). Prices on the market can differ from one supplier to another. Some PVC pipes are sold at 0.80€ per meter. The cost of a bend would be between 0.40€ to 0.80€.

All in all, we find the following:

- 2 pipes per slab: 17 000m of pipes, 299 bends ⇒ 3 400€ to 15 000€
- 3 pipes per slab: 27 000m of pipes, 449 bends ⇒ 5 600€ to 22 000€

It is clear that placing 3 pipes rather than 2 inside a 200mm slab clearly demands an additional expense of more than 50%. However, we will consider this option in the context of control in a future work, to determine whether the higher installation price is balanced by higher performance and lower maintenance expenses.

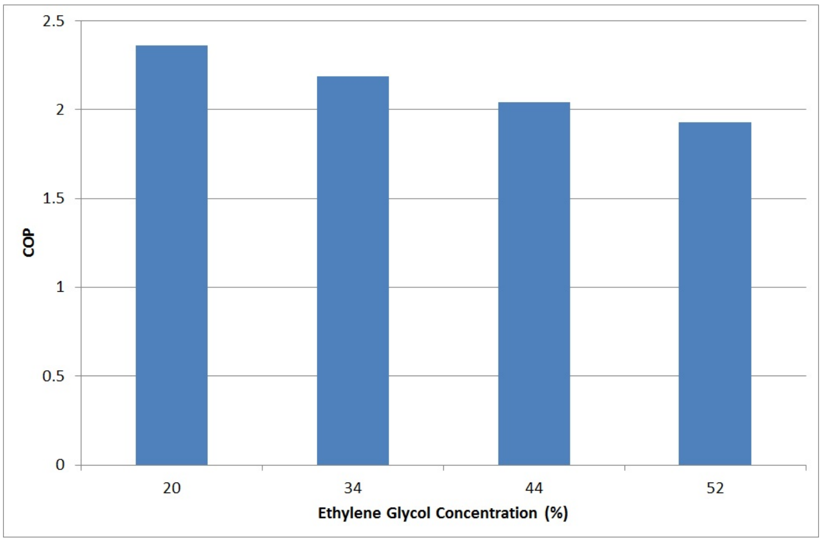

Figure 16: The COP as a function of the ethylene glycol concentration.

7.4. Cost of the refrigerants: Ethylene Glycol vs. Ammonia

The price of liquid Ammonia seems to be lower than the price of Ethylene Glycol, even if a serious increase of the costs of liquid Ammonia occurred some time ago. Today liquid Ammonia is sold indeed from \$500 to \$800 per ton (approx. 400€ to 600€), whereas Ethylene Glycol is sold at \$800 to \$1 400 per ton (600€ to 1 000€).

Liquid Ammonia used to have the lowest price compared to all the other secondary refrigerants. For 1 kg, it was 5 to 8 times cheaper than others. It is also the lightest fluid used as a refrigerant, that makes it cheaper to use (from 11 to 17 times w.r.to. Ethylene Glycol). After the rise of prices, it is still 1.5 to 2 times less expensive (average price in April 2012: \$500 per ton, ≈ 380€).

Moreover, even though the concentration of the refrigerant certainly determines the price, it is unlikely that in our case the use of Ethylene Glycol turns out to be more convenient than Ammonia.

Overall liquid Ammonia seems to be *the best solution* (better COP_{sys}, many advantages such as chemical neutrality with pipes, olfactory detection of leaks, no green-house effect contribution and so on).

7.5. Ethylene Glycol and Ammonia Concentrations

According to the literature, excessive glycol concentrations should be avoided if we seek to maximize the heat transfer rate. Moreover glycol concentrations less than 20%vol shall not be used, since it is a nutriment source for bacteria at these low rates [26]. We also know that some existing cooling systems have the following features: brine at 34%vol Mono Ethylene Glycol, brine at 38%vol Mono Propylene Glycol, Liquid Ammonia at 17-18%vol. Generally a cooling antifreeze liquid has about a 40%vol concentration.

It is reported in Table 2 that for a -8°C freezing point (average pipe temperature in our data) the minimal concentration is 18,1%vol. This increases to 52%vol for -39°C [27].

To sum up, ideal Ethylene Glycol concentrations should be in any case between 20 and 52%vol.

We are more likely to suggest using liquid Ammonia rather than Ethylene Glycol: another plea for this choice is that NH_3 can be used with a lower concentration, which makes it even cheaper.

Density (g/cm^3) @20°C	%vol	Freezing Point (°C)
1.024	18.1	-8
1.030	22	-10
1.067	49.6	-39
1.072	53.7	-45

Table 2: Ethylene Glycol concentrations and corresponding freezing points

8. Conclusions and outlook

In this paper we have outlined a functional method to study the optimization of cooling in ice hockey halls, and provided the optimal brine fluid and concentration for an average-sized ice rink in standard operating conditions.

Investigating the COP of the *entire* cooling system as a function of several variables is an interesting option, as it provides with precise prescriptions on the system design that do not demand expensive and time-consuming preliminary tests.

This *a priori* approach makes it possible to quickly address questions such as: the weight of volumetric flow and brine characteristics on the refrigerator performance; the role of the pumping power in the total electric consumption; choosing the most efficient and cost-balanced brine.

We find that the pumping power W_p corresponds to $\sim 7\%$ of the total electric consumption, and that the volumetric flow is uninfluent on the system COP$_{sys}$ for any technically acceptable values.

Accordingly, we show that energy optimization should not be achieved by acting only on the pumping power, since a much more crucial role is given by the other sources of electric consumption such as the chiller, heat exchanger and condensers. Choosing the optimal brine fluid is also very relevant to achieve a high COP$_{sys}$.

By using the method of functional optimization, we have then singled out which brine fluid is optimal for the configuration at hand. It is shown that COP$_{sys}$ is mostly sensitive to the *specific heat* of the fluid. Moreover, there is a specific value of the *density* for which the COP is maximal (Figure 18), regardless of the specific brine chosen.

It follows that ammonia is a better choice than ethylene glycol, and that glycol at concentration between 20% and 34% is to be preferred. We have compared these predictions with technical contraints, and a cost estimation is provided in Section 7.

Design possibilities regarding the pipe size and depth inside the concrete slab have been investigated as well. Transient heat transfer in a two-dimensional portion of the slab has been simulated with COMSOL Multiphysics, to give a consistent check to our assumptions at a qualitative level.

We confirm rigorously the well-known practical result that the pipe size and depth do not enhance appreciably the process. However, increasing the number of pipes provides with a more uniform temperature profile at the ice surface. This might be an interesting option to consider when assessing control methods of the brine flow.

Our predictions are obtained with a model that is validated with measured data obtained at an ice rink in Finland, and are compared to technical and cost-related constraints.

We finally stress that the multivariable approach here applied is general, and as such it can be applied to the rigorous study of diverse situations in engineering.

9. Acknowledgements

The authors would like to thank Kirsi Säkkinen for the measurements of the brine properties, and acknowledge support by the Academy of Finland.

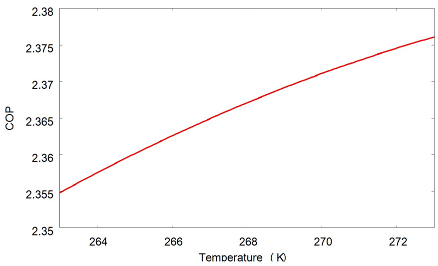

Figure 17: The COP in function of the mean brine temperature only.

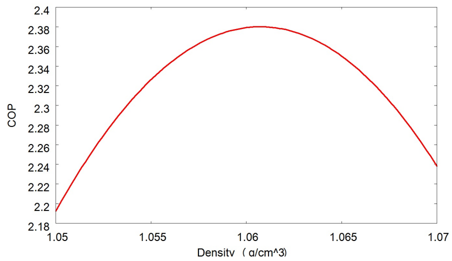

Figure 18: The COP as a function of the brine density ρ.

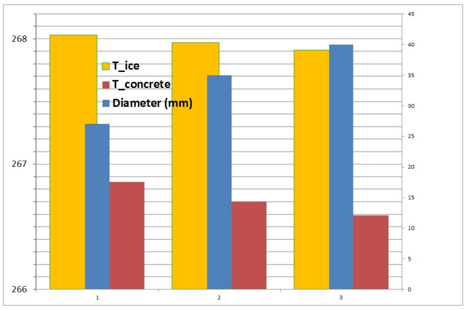

Figure 19: Temperatures for pipe diameters $27\,mm$, $35\,mm$ and $40\,mm$.

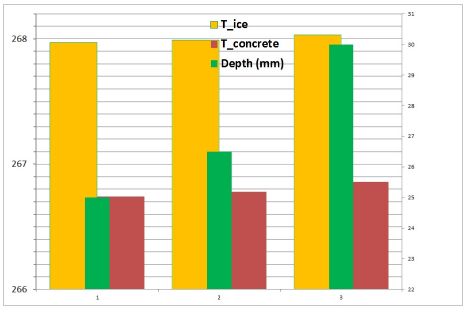

Figure 20: Temperatures for pipe depths $25\,mm$, $26.5\,mm$ and $30\,mm$.

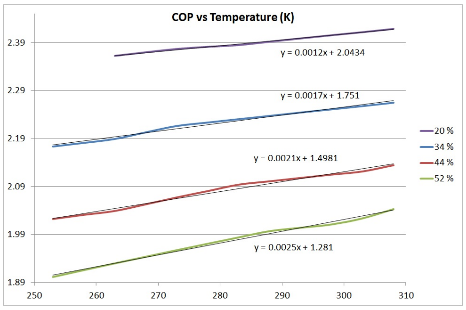

Figure 21: The COP as a function of the brine temperature for several ethylene glycol concentrations.

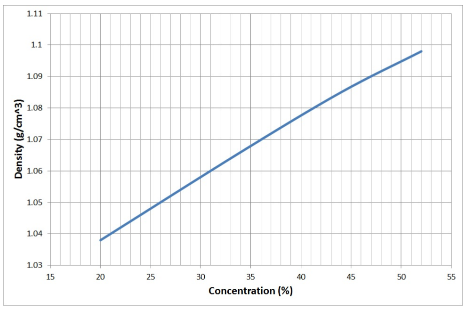

Figure 22: Density of ethylene glycol as a function of concentration.

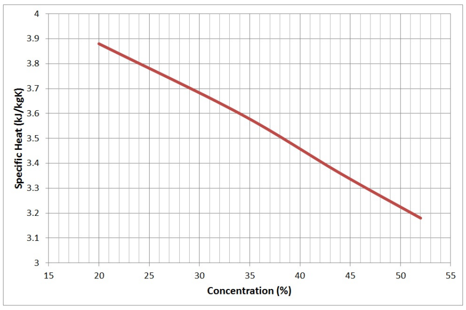

Figure 23: Specific heat of ethylene glycol as a function of concentration.

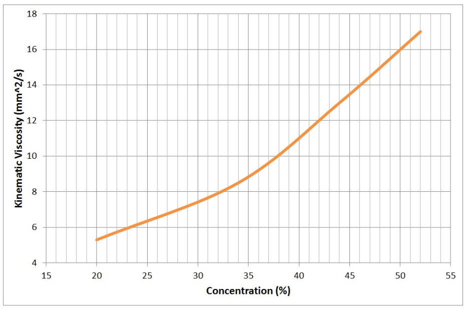

Figure 24: Kinematic viscosity of ethylene glycol as a function of concentration.

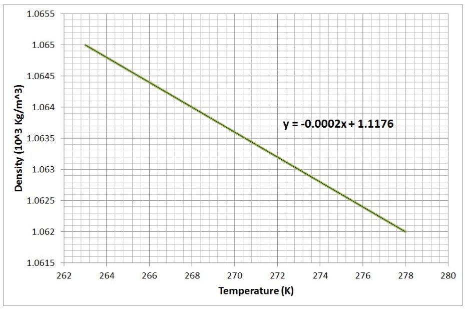

Figure 25: Density of an ethylene glycol solution at 34% in function of temperature.

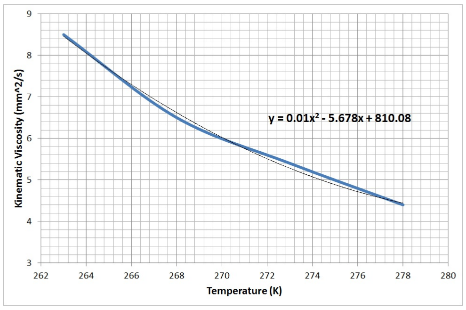

Figure 26: Kinematic viscosity of ethylene glycol at 34% in function of temperature.

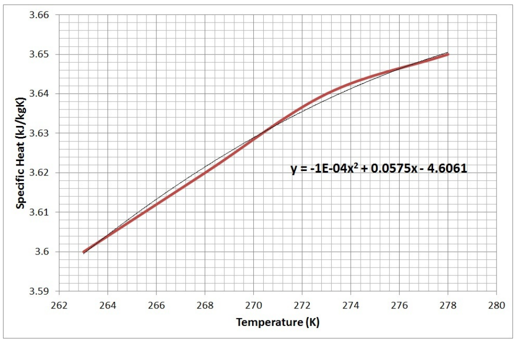

Figure 27: Specific heat of an ethylene glycol solution at 34% in function of temperature.

References

[1] Antuan Negiz, Mehmet A. Hastaoglu, and Robert A. Heidemann. Three-dimensional transient heat transfer from a buried pipe. *I. Laminar flow, Chem. Engin. Science*, 48:3507–3517, 1993.

[2] Mo Chung, Pyung-Suk Junga, and Roger H. Rangler. Semi-analytical solution for heat transfer from a buried pipe with convection on the exposed surface. *International Journal Of Heat and Mass Transfer*, 42:3771–3786, 1999.

[3] Gabriel Teyssedou, Radu Zmeureanu, and Daniel Giguere. Thermal response of the concrete slab of an indoor ice rink (rp-1289). *HVAC & R Research*, 15, May 2009.

[4] Omar Bellache, Mohammed Ouzzane, Roberto Sunyé, and Nicolas Galanis. Transient Energy Flux Analysis in an Indoor ice Rink Using a CFD Approach. Technical report, CANMET Energy Technology Centre, Jan 2008.

[5] Mazyar Karampour. Measurement and modeling of ice rink heat loads. Master's thesis, Royal Institute Of Technology, Stockholm, Sweden, 2011.

[6] Pavel Makhnatch. Technology and Energy Inventory of Ice Rinks. Master's thesis, Royal Institute Of Technology, Stockholm, Sweden, 2011.

[7] Omar Bellache, Mohamed Ouzzane, and Nicolas Galanis. Numerical prediction of ventilation and thermal processes in ice rinks. *Building and Environment*, 40(3):417–426, 2005.

[8] Lofti Seghouani, Ahmed Daoud, and Nicolas Galanis. Prediction of yearly energy requirements of indoor ice rinks. *Energy And Buildings*, 41:500–511, 2009.

[9] Hakan Caliskan and Arif Hepbasli. Energy and exergy analyses of ice rink buildings at varying reference temperatures. *Energy and Buildings*, 42(9):1418–1425, 2010.

[10] Waqas Khalid. Energy usage prediction model comparing indoor vs outdoor ice rinks. Master's thesis, Royal Institute Of Technology, Stockholm, Sweden, 2012.

[11] Miska Räikkönen. *Jäähallin energiatehokkuus (Diploma thesis, in Finnish). Aalto University, Espoo, Finland.* 2012.

[12] Editor: Å. Melinder. Handbook on indirect refrigeration and heat pump systems. Technical report, Dept. of Energy Technology, KTH.

[13] Kumpati S. Narendra and Snehasis Mukhopadhyay. Adaptive control using neural networks and approximate models. *IEEE Transactions on Neural Networks*, 8(3):475–485, May 1997.

[14] Kumpati S. Narendra and Kannan Parthasarathi. Identification and control of dynamical systems using neural networks. *IEEE Transactions on Neural Networks*, 1(1):4–26, Mar 1990.

[15] Thomas Parisini, Marcello Sanguineti, and Riccardo Zoppoli. Nonlinear stabilization by receding-horizon neural regulators. *International Journal Of Control*, 70(3):341–362, March 1998.

[16] Thomas Parisini and Riccardo Zoppoli. Neural networks for feedback feedforward nonlinear control systems. *IEEE Transactions on Neural Networks*, 5(3):436–449, May 1994.

[17] Thomas Parisini and Riccardo Zoppoli. Neural approximations for multistage optimal control of nonlinear stochastic systems. *IEEE Transactions on Automatic Control*, 41(6):889–895, Jun 1996.

[18] Jonas Sjöberg, Qinghua Zhang, Lennart Ljung, Albert Benveniste, Bernard Deylon, Pierre-Yves Glorennec, Håkan Hjalmarsson, and Anatoli Juditsky. Nonlinear black-box modeling in system identification: a unified overview. *Automatica*, 31:1691–1724, Jun 1995.

[19] Riccardo Zoppoli and Thomas Parisini. Learning techniques and neural networks for the solution of N-stage nonlinear nonquadratic optimal control problems, in Systems, Models and Feedback. *IEEE Transactions on Automatic Control*, 41(6):193–210, Jun 1992.

[20] Riccardo Zoppoli, Marcello Sanguineti, and Thomas Parisini. Approximating networks and extended Ritz method for the solution of functional optimization problems. *Journal Of Optimization Theory and Applications*, 112(2):403–440, Feb 2002.

[21] Alexander Cohr Pachai. Energy consumption and heat recovery of refrigeration system in modern arena. *Johnson Controls Building Efficiency*, 2011.

[22] Natural Resources Canada. Optimization of ice and concrete slab thickness. Technical report, CANMET Energy Technology Centre, 2003.

[23] C.J Hewetson. Ice Skating Rink Design Considerations, Feb 2003.

[24] René Fasel. Technical guidelines of an ice rink. Technical report, International Ice Hockey Federation, 2011.

[25] Benoit Lazure. Technical fact sheets on the impacts of new energy efficiency technologies and measures in ice rinks. Technical report, The Association des Arénas du Québec, 2003.

[26] G.F Yuzwa. Converting a closed-loop water system to a glycol system. In *Alberta Public Works, Supply & Services Property Management*, 1999.

[27] William M. Haynes. *CRC Handbook of Chemistry and Physics, 92nd Edition.* June 2011.